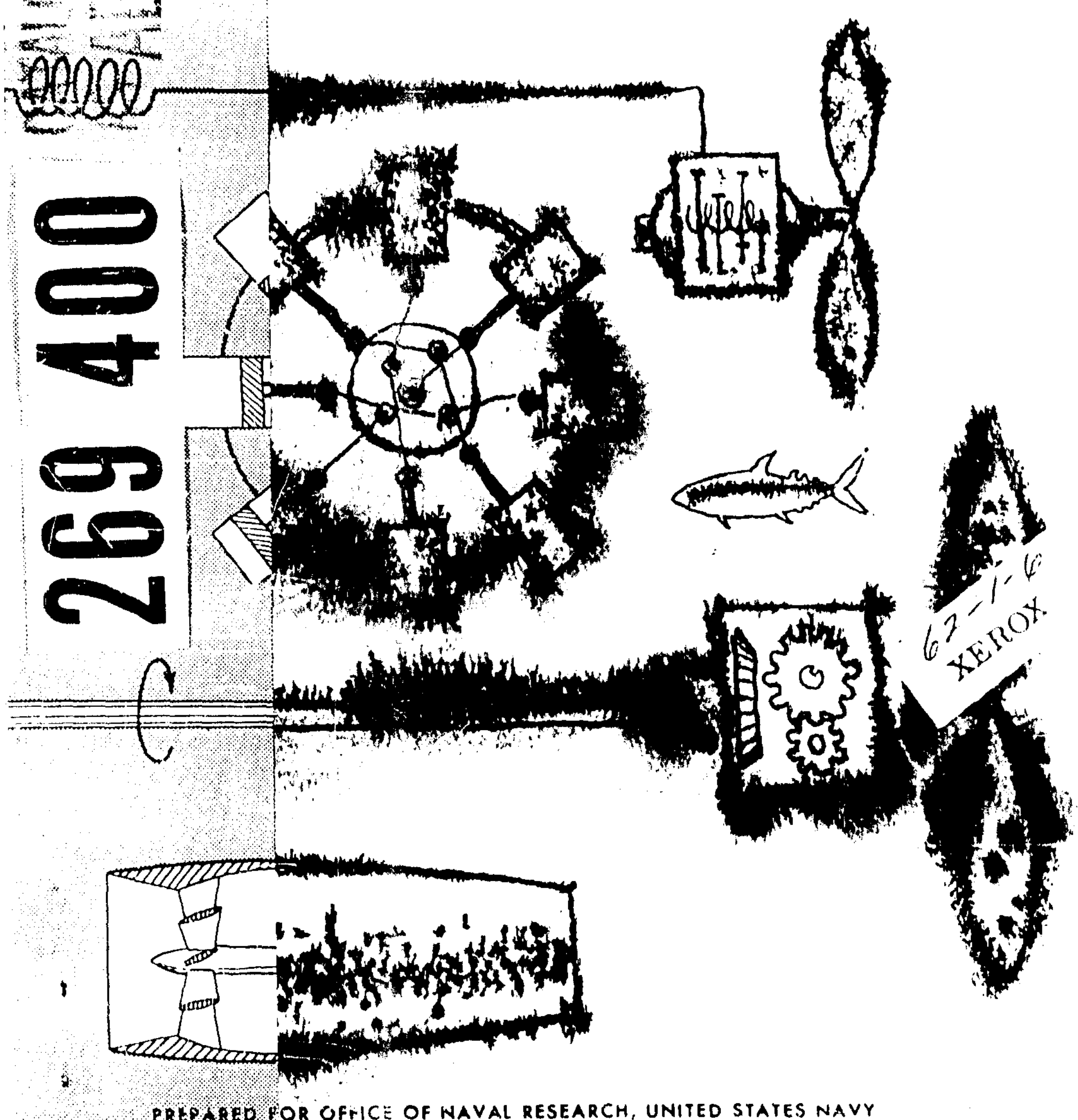
APPENDIX 1

APPRAISAL OF VARIOUS MECHANICAL TRANSMISSIONS

H. STERN

LOW MAINTENANCE MACHINERY FOR SUBMARINE POWER PLANTS



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PREPARED FOR OFFICE OF NAVAL RESEARCH, UNITED STATES NAVY
BY MEDIUM STEAM TURBINE, GENERATOR AND GEAR DEPARTMENT
AND THE GENERAL ENGINEERING LABORATORY

GENERAL ELECTRIC

Appendix 13

APPRAISAL OF VARIOUS MECHANICAL TRANSMISSIONS

By

H. Stern

October 1961

This report contains a detailed discussion of one phase of the Low Maintenance Machinery Study program performed by the General Electric Company under Contract NONR 3485 (00) and was written at the conclusion of that phase. It is presented in support of the Final Report of December 1961. While the results of this study phase contribute to the findings of the Final Report, certain conclusions drawn in the Final Report have the benefit of additional information obtained or generated in the over-all program and subsequent to the writing of this Appendix. The conclusions presented herein are, therefore, subject to modification by the Final Report.

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SUPPLEMENT B

Appraisal of Various Mechanical Transmissions

by
H. Stern

This phase is concerned with the appraisal of various mechanical means of transforming thermal power into hydraulic power, i.e., steam energy into ship thrust. The conclusions drawn from this work are twofold.

1. The steam motor direct drive for a propeller makes a very attractive transmission. It is being explored in detail under a separate phase of the program.
2. The torque converter development, once sponsored by the U. S. Navy, could result in substantially improved propulsion systems with either turbine or reciprocating steam prime movers. Further development should not be neglected. Presently, efficiency reliability, and size are not sufficiently perfected to offset the advantages of quiet operation and maneuverability.

Other mechanical transmission schemes which were investigated include:

1. Hydraulic split transmissions using both hydrokinetic and hydrostatic elements.
2. Belt drives.
3. Variable pitch propellers.

Hydrodynamic shear drives are being evaluated under a separate program phase. Of the transmission currently thought feasible, the Fottinger transformer and the steam motor direct-drive offer the greatest incentive for further investigation.

The following sections discuss details of the various study phases:

1. Basic Power Transfer

The problem under consideration is the load match between the primary energy source of an atomic submarine and the thrust generator of the ship. Saturated steam is the fluid at the energy source; pressure ranging from 300 psi at full load to 600 psi at no load are encountered. The water surrounding the ship is the only logical thrust generating fluid to be considered. Since no ship speed and thrust data are furnished by the ONR, I shall assume values which appear reasonable and will lend physical meaning to the thoughts presented here. A maximum ship velocity of 60 ft/sec is chosen with a corresponding ship drag of 137,000 lbs.

Basically, the problem converges on the fact that the steam energy, when converted into such Kinetic energy that is useful for the generation of force or torque, becomes available to us at velocities of the order of 10^3 to 10^4 ft/sec. The propulsion mechanism has to generate force from Kinetic energy at velocities of 10 to 100 ft/sec; that is two orders of magnitude below the velocity of the steam.

From these considerations, we can deduce the desirable physical features of a direct steam/water propulsion mechanism such as proposed by R. J. Hooker (Ref. 7). The relations are shown in Figure 1, using the Kinetic energy of steam. If 80% efficiency is assumed in both conversion steps, we can postulate a "rowing machine" whose over-all efficiency is 64% (not counting mechanical losses), whose physical dimensions are shown in Figure 2a. For higher efficiency the "lever" ratio of 19:1 must be raised, so that at 36:1 a total efficiency of 81% results. It so happens that this approach yields numbers, which in spite of their obvious crudity are reminiscent of actual trends and size ratios in ship propulsion systems today. The graph shows that the optimum size of 4-1/2 ft^2 of the "steam paddle" occurs when both steam and water side are 85% efficient and the lever ratio K between them is about 27:1. As the lever ratio is reduced the efficiency drops also, while the "steam paddle" area rises sharply. In all instances, the velocities of the "steam paddle" are such as to be unthinkable in terms of reciprocating motion. The use of the turbine wheel and gear transmission obviously is the "cleanest" and most direct translation of these theoretical, idealized machines into actual practical mechanisms. Considering that orders of magnitude only have been used in these computations, the similarity in size of steam turbine admission area vs. "paddle" areas, of gear ratios vs. level ratio, and of "water paddle" area vs. propeller area is striking.

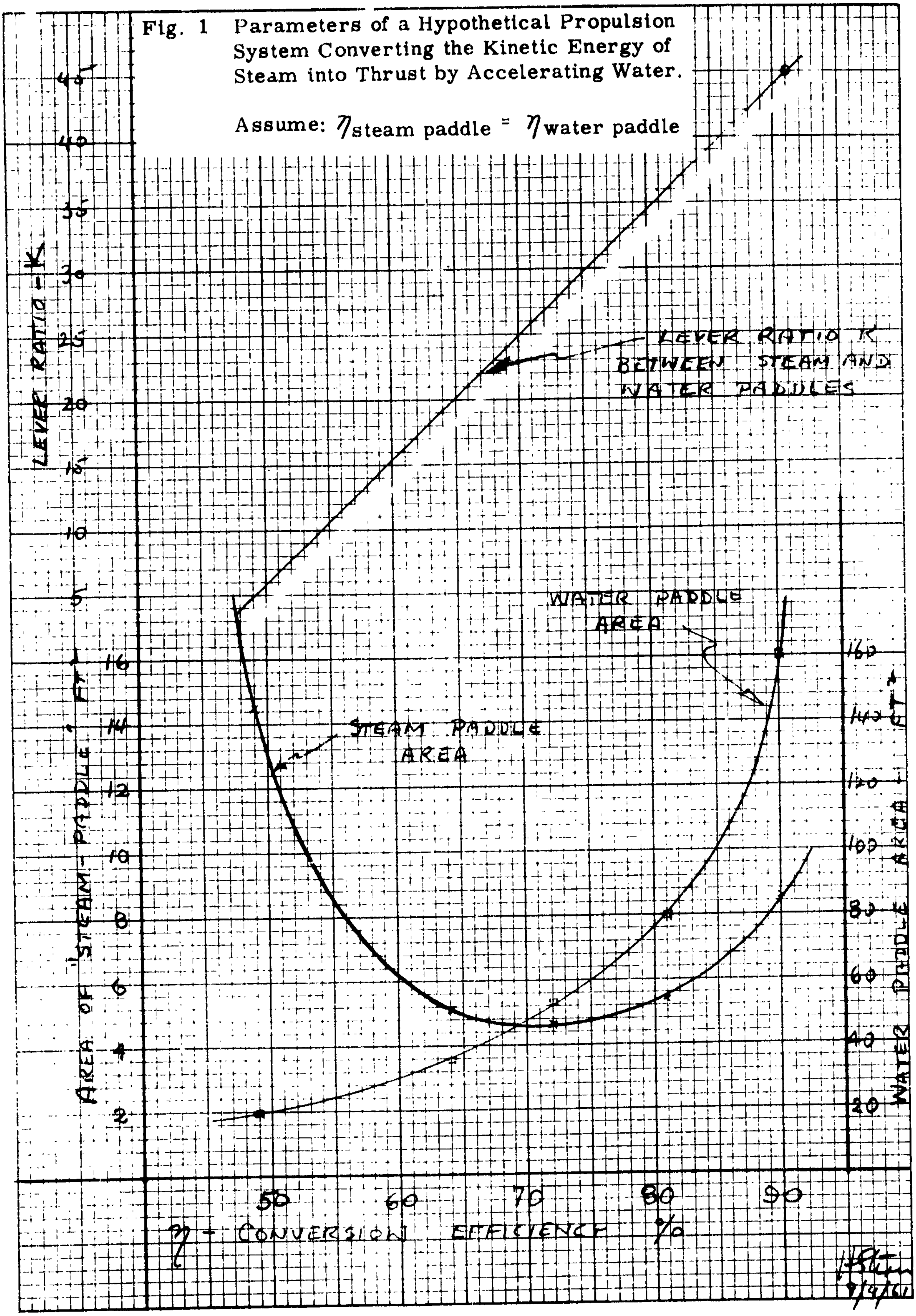
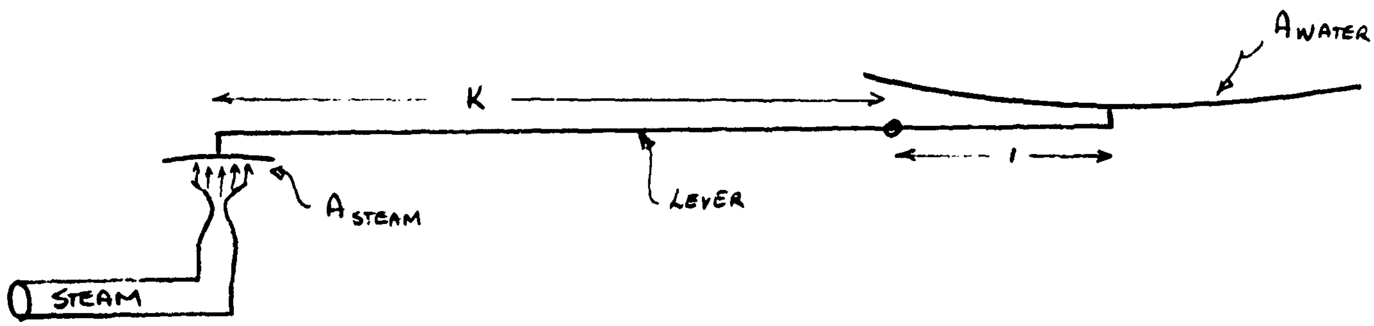


Fig. 1 Parameters of a Hypothetical Propulsion System Converting the Kinetic Energy of Steam into Thrust by Accelerating Water.

Assume: $\eta_{\text{steam paddle}} = \eta_{\text{water paddle}}$

Fig. 2a Schematic Diagram of Hypothetical Rowing Machine Energized by Kenetic Energy of Steam.



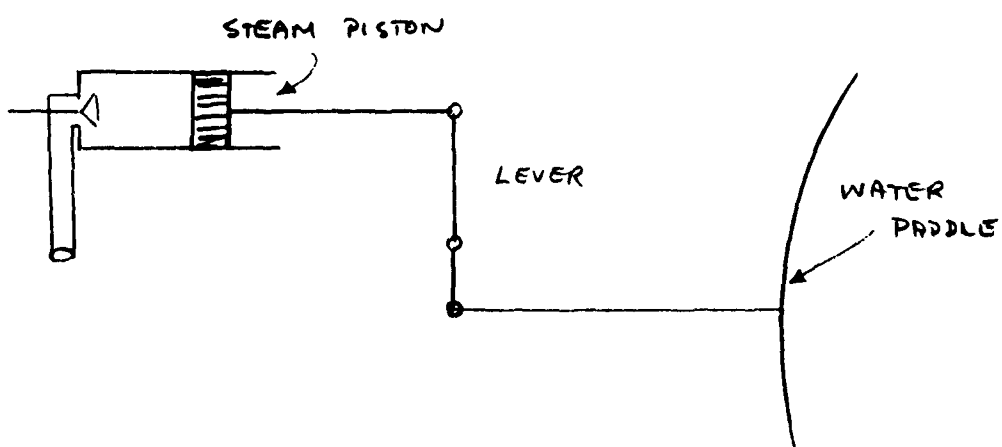


Fig. 2b Schematic Diagram of Hypothetical Rowing Machine Energized by Pressure-Volume Energy of Steam.

Alternately, we can devise a "rowing machine" operated by steam not using the kinetic energy, but rather the flow and expansion energy. At a mean effective pressure of say 100 psi, we could attain steam piston areas of around 10 - 12 ft^2 area traveling at 70 - 90 ft/sec velocity. If coupled through appropriate linkages to the "water paddle" these velocities can be reduced to the commonly used piston speeds of 20 - 30 ft/sec. (Fig. 2b)

In both the foregoing cases, the water paddle was moving intermittently at more than ship velocity. Clearly the rotating propeller is to be preferred to the reciprocating paddle or piston, particularly when speeds between 70 and 90 ft/sec are necessary. Since a fairly wide range of rotating propeller speeds can satisfy the conditions required of the "water paddle", we have before us a justification for what we already know - that a turbine gear, propeller combination is an effective and versatile power transmission system between steam and water. We also can deduce that with only minor modifications (a 3 or 4:1 lever) the steam piston can be coupled to the "water paddle" which we already have transformed into a rotating paddle or propeller. By this argument the steam motor becomes a valid replacement for both turbine and gear, since it can incorporate the appropriate lever ratio in its crank shaft to enable normal speed pistons to work directly on the propeller shaft.

The foregoing examples are analogous to fixed ratio transmissions. Considerable advantage can be gained in using variable ratio drives. The significant factor in the use of a variable ratio transmission is the fact that it permits the prime mover to impart more nearly full power to the propeller under various ship maneuvering conditions. A fixed ratio transmission can only supply that power which corresponds to the prime mover torque at a given propeller speed. A plot of such conditions is shown in Figure 3 taken from Ref. 2. Thus aside from the ability of the prime mover to operate at nearly constant speed at all times, a variable ratio transmission also permits faster maneuvering.

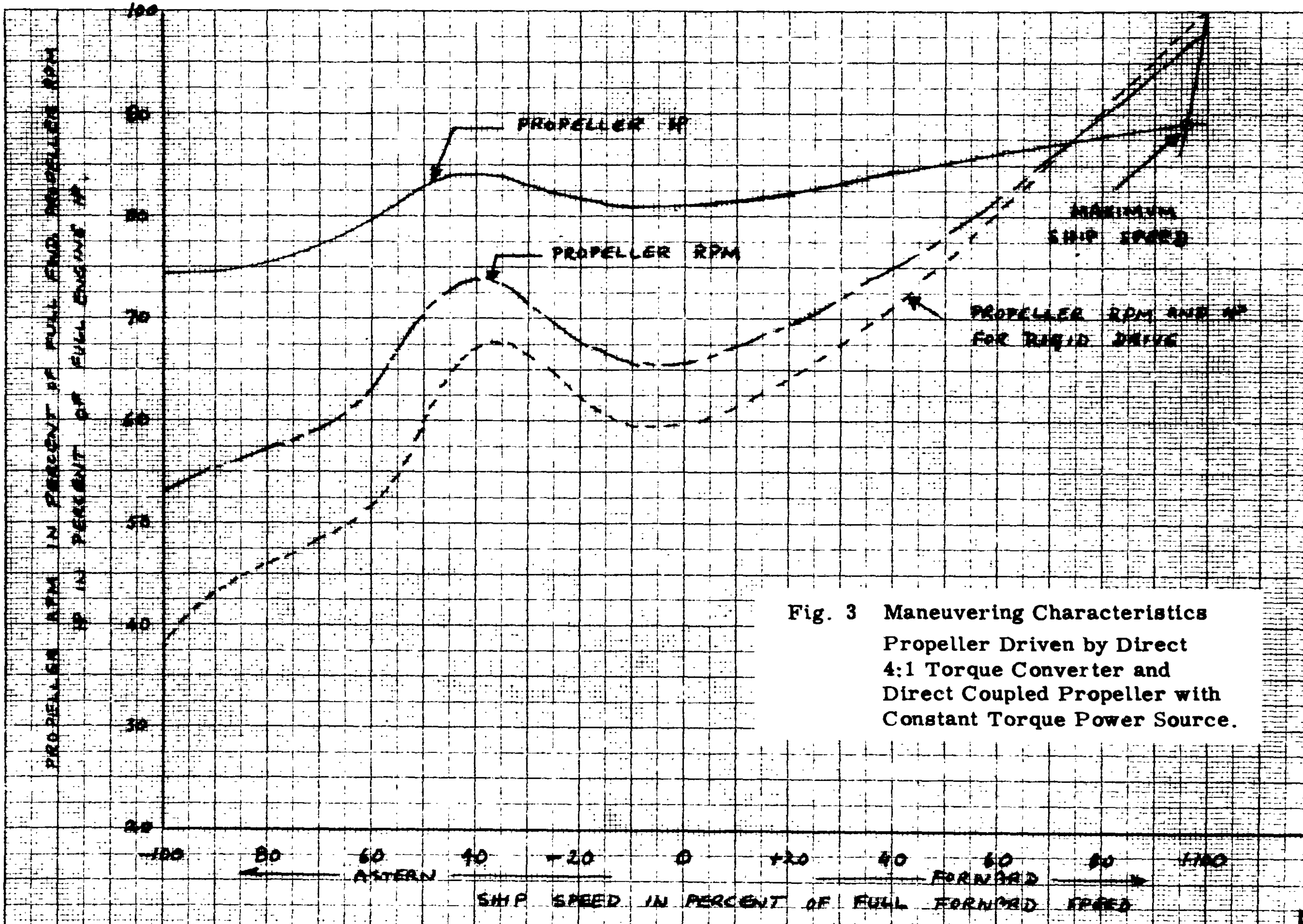


Fig. 3 Maneuvering Characteristics Propeller Driven by Direct 4:1 Torque Converter and Direct Coupled Propeller with Constant Torque Power Source.

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2. Transmission Evaluation

Re-examination of present practice of geared turbine drives brings out the facts which cause us to question their desirability on the one hand and appreciate their value on the other. Noise, weight, and size are the objections raised to gearing. To offset these, reliability, efficiency and cost are the features which to date have always helped retain geared transmissions in the top position among nuclear submarine drives.

There are many ways open to bridge the 30:1 speed gap between an efficient steam turbine and the propeller. A listing of those investigated within the frame of this program is shown in Table 1. Both gearless and partially geared mechanical transmission systems are feasible.

2.1 Hydrodynamic Transmissions

Examination of the fluid torque converter or Fottinger transformer reveals the following possibilities. Bendersky (Ref. 1) goes into gear-torque converter combinations, each of which yields approximately a 5:1 torque multiplication. We can immediately reject the combination of high speed gear and conventional speed T-C. It retains the noise of the gear transmission. While the high speed T-C with the low speed gear may be somewhat quieter, it adds the development problem of a 6,000 - 10,000 rpm torque converter whose efficiency is highly questionable at this state of development. It still retains a noisy gear train and all the weight and bulk of low speed gears. The double reduction T-C proposed by Benderksy (Ref. 1) suffers from lower efficiency as well as complexity. This leaves two further alternatives: a) Reduce turbine speed to permit a single reduction conventional T-C, or b) Develop a high torque ratio converter of 30 to 1. A compromise between these two alternatives is offered in the (Ref. 2) as a 10:1 torque multiplier suitable for I.C. engine transmissions or for coupling low speed turbine to high speed propeller.

Alternative a) involves a turbine drive of 1,000 or 2,000 rpm for maximum propeller speeds of 200 and 400 rpm respectively. The torque converter for such applications has been developed in somewhat smaller sizes than that demanded by the current specifications. Ref. 3 - 5 described a 1250 hp 4:1 converter, built by National Supply Co. and tested for noise and performance at the E.E. Sta., Annapolis. This unit, according to the reports available, failed to meet expectations in several respects. The seals consistently caused trouble not only by leakage but also through excessive wear. Even a complete redesign did not eliminate these problems. Oil from the converter and the bearings mixed. Filters clogged with seal debris. The unit exhibited much lower efficiency than predicted, top performance being about 65% at 4:1 torque multiplication and 800 rpm. The performance trends appear to indicate that better efficiency could have been obtained at a speed or hp level higher than that for which the machine was designed. Even if the speed was raised by 40%, doubling the power of the transmission, it does not appear likely that this machine would have done better than 70% at 4:1 torque ratio. The third defect of the transmission appears to have been the noise level in the auxiliaries. While Ref. 5 stressed the quiet performance of the T-C itself, it says that the circulating pumps had to be removed and separately mounted and that all fluid connections to the T-C were made with rubber hose. My conclusions from these reports run as follow:

1. A T-C can be designed to overcome the simple mechanical defects of seal leakage and galling exhibited by the unit tested.
2. A T-C must be designed to use the same fluid in all its circuits. The T-C using a 5W oil for the power loop, a marine hydraulic fluid for the control actuation, and possibly a heavier marine bearing lube oil for bearings does not make for a well designed reliable mechanism. This was proven by the test failure.
3. A T-C should drive its accessories from the main power input shaft. This will permit low rpm pumps with correspondingly low noise level and will yield higher reliability then independently driven high speed pumps. A single fluid system will require only one pump at most; possibly the impeller of the T-C itself can be used to furnish both control and lube oil pressure.
4. Refs. 1 and 2 stress that efficiencies in the 80 - 90% range are attainable with a 4:1 machine. Refs. 3 and 4 mention improved fluid dynamic design. Even the smallest T-C's obtain better efficiency than the National Supply Co.'s test unit.

Further development work is indicated to realize the potential predicted by experts for the torque converter in the large size range. The reports cited as References 2, 4 and 5 indicate that the test unit was intended for noise evaluation and justified the hopes of the designers in this respect. The tests appear to have been terminated because of undue mechanical difficulties encountered with the specific machine that was furnished for test. The low noise level, the possibility for variable transmission ratios, and the high reliability, demonstrated in other applications, are added to increase ship maneuverability. Currently, the over-all effect of efficiency for several ship duty cycles is being evaluated in terms of steam consumption and machinery bulk. All evidence to date points to continuing work on this type of transmission for immediate applications to submarines.

A scaled-up diagram of the National Supply Co.'s torque converter is shown in Figure 4. The possibility that this unit may be satisfactory for coupling a high speed steam motor drive to a 200 rpm propeller is being investigated.

Alternative b, the 30:1 torque converter is discussed in Ref. 1. It consists of a multistage turbine driven by an axial flow pump.

This high speed ratio torque converter appears to have a considerable size advantage stemming from its 3 ft. diameter turbine rotor. To date, no information is available on the potential efficiency of such a machine. There is no indication whether a reasonable efficiency can be attained within any mechanically feasible configuration. Unlike for the low ratio torque converter, no test data on high ratio machines of similar design is available. For the current program, a dual development is necessary to make a high ratio torque converter of this type become a reality. The first part of this program would be to determine the feasibility of the design in principal, and build and test the machine to verify the performance. Secondly, a development program would be necessary to convert the feasibility prototype into a production machine.

The compromise torque converter design explored in the National Supply Co.'s proposal Ref. 2, appears to offer much the same course of action as the high ratio torque converter of Mr. Bendersky (Ref. 1). While the degree of risk may be lower in the 10 to 1 torque converter, and some of the early design calculations may have been performed, such a machine is still to be built, tested and developed. In addition, however, it will be necessary to develop a turbine that can operate efficiently at the low rpm. Consequently, I view both of these alternatives of

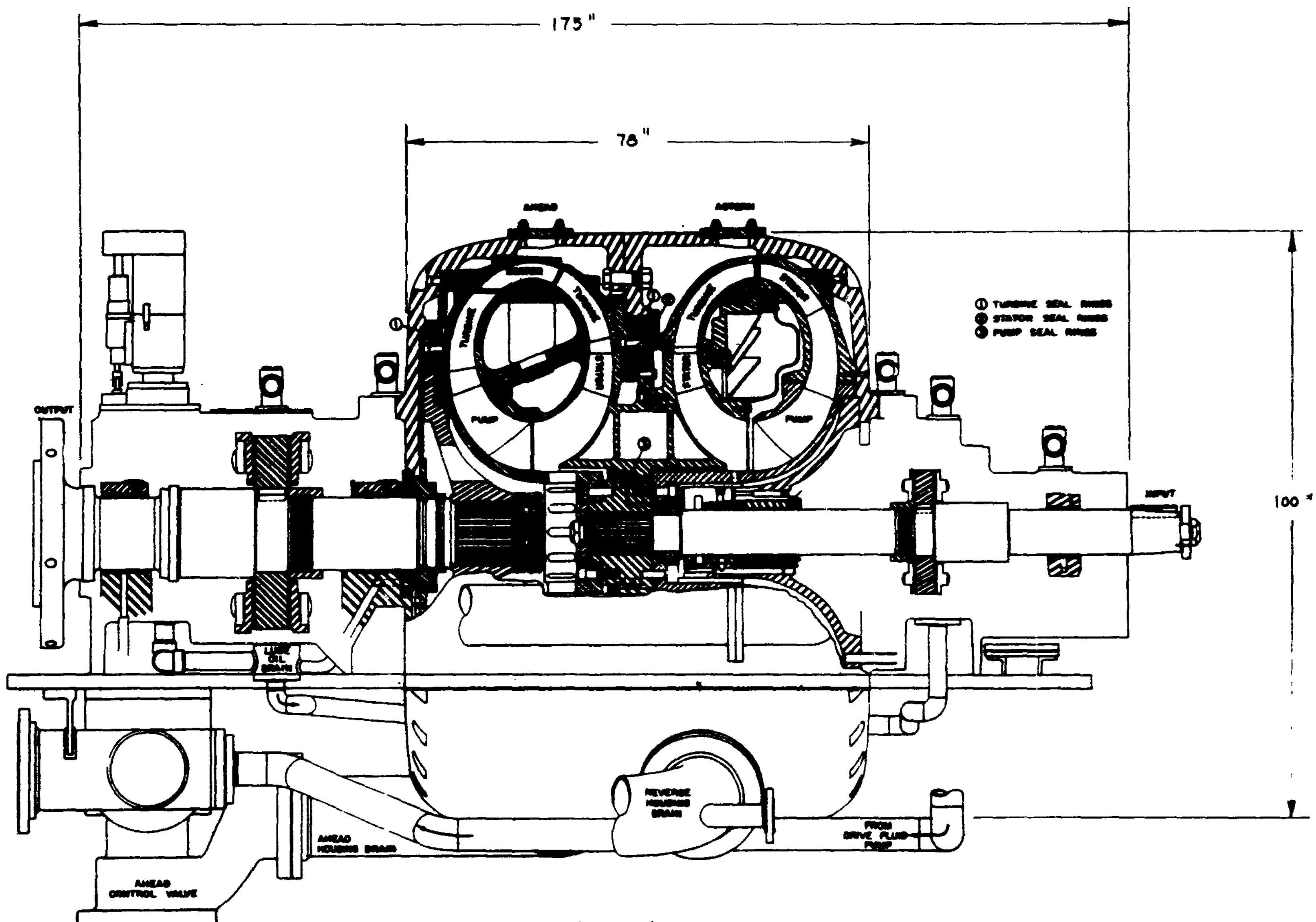


Figure 4

National Supply Company

Torque Converter Cross-Section with Dimensions Scaled up to Approximately 15,000 HP at 1,000 RPM Input Speed

Adapted from Reference 3

low potential for the presently envisioned program but recommend careful exploitation of their possibilities for future reference.

2.2 Hydraulic Transmissions

In addition to hydrodynamic transmissions, hydrostatic transmissions were examined as reported in Ref. 8. Given a high speed prime mover such as a steam turbine, it was necessary to replace the conventional hydraulic piston pump with a corresponding multi-stage turbine because the high noise level of a high speed piston pump could not be tolerated in a submarine drive. The motor on the other hand is operated at a maximum speed of 200 rpm can be built to be relatively noiseless. At the low speeds encountered here noise reduction is achieved in such a machine by careful scheduling of the inlet and exhaust valving of each cylinder. The advantages of such a hydrostatic drive motor would be very rapid reversibility without the use of a separate reversing unit. It is a compact transmission, readily deployable in many spacial configurations and is light in weight, see Figure 5a. A hydraulic motor of the torque range required for each of two propellers is outside current manufacturing practices but represents no significant engineering advance beyond normal design development. A hydraulic motor of this type, requires a multi-stage axial flow pump. Transmission efficiencies are expected to be essentially those of the pump and its defuser, see Figure 5b. The hydraulic motor itself could have an efficiency of 93 to 95% which is almost constant over the entire speed range. The use of hydraulic fluid is necessary in such a transmission.

The possibility exists for developing a fully submerged sea water fluid drive for the propellers. Since no such equipment has been built to date, considerable engineering design and development will be required to bring such a machine within the realm of a product reality. Such a transmission is briefly discussed in Ref. 8 also. Its vulnerability to marine fouling and battle damage would have to receive careful attention.

2.3 Mechanical Drives

V belt or rope transmissions were considered for the purpose of comparison in size to conventional gear drives connecting the turbine to the two propeller shafts. Such a V belt drive would have very high reliability because of the large number of V belts which operate in parallel. The efficiency of such a drive would be of the same order of magnitude as that of a gear transmission. It would be expected to operate without lubricant fluid. Cooling of the V belt sheaves would be accomplished by internal circulation of water or low pressure steam. This coolant could be confined to the small diameter sheave where the greatest amount of slip and most of the

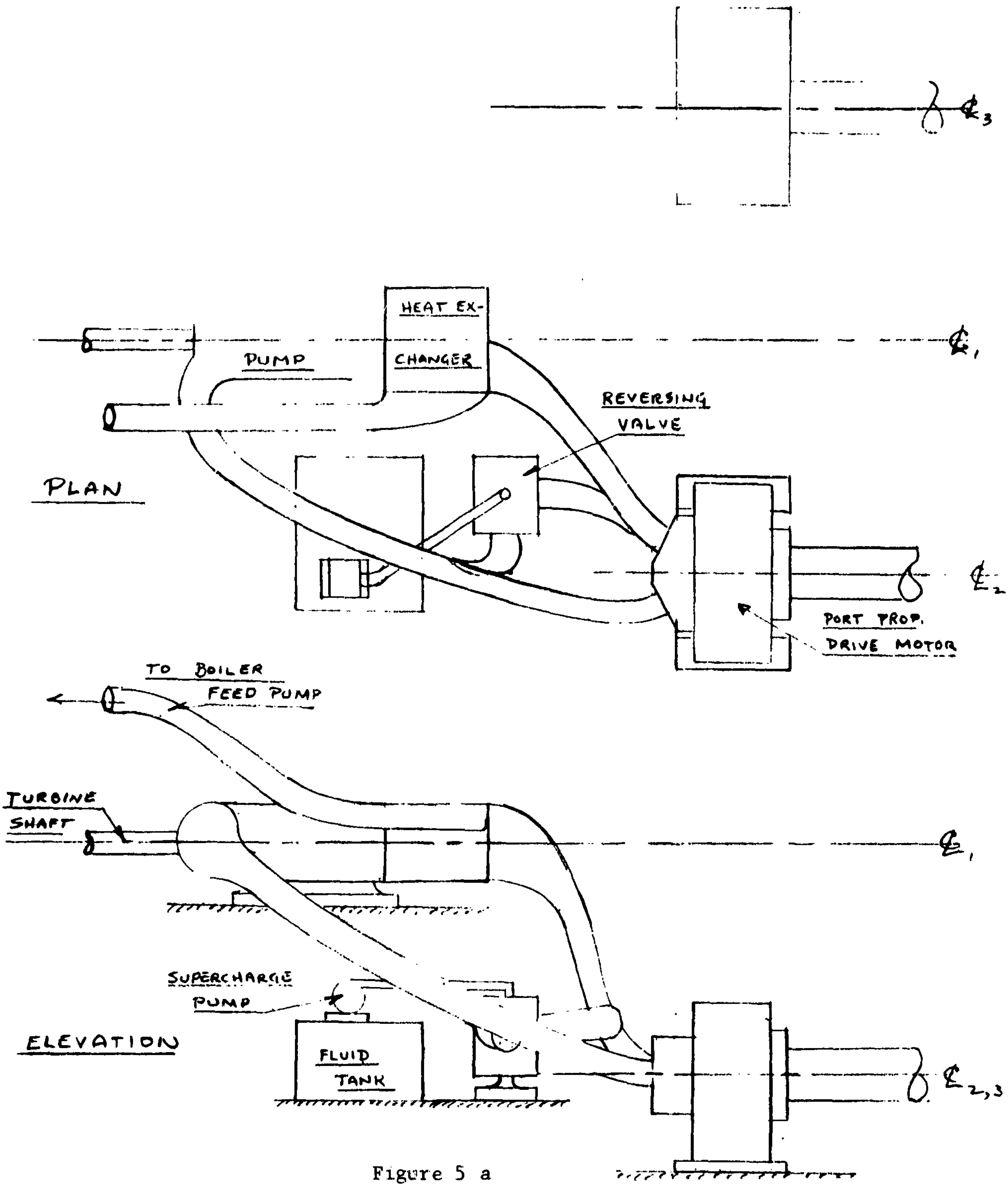


Figure 5 a

Schematic Layout of Split Hydraulic Transmission

Total 15,000 HP Maximum Output at Two 200 RPM Radial Piston Propeller Drive Motors.

5,000 RPM Multistage Axial Flow Pump.

SCALE :
0 1 2 3 4 FEET

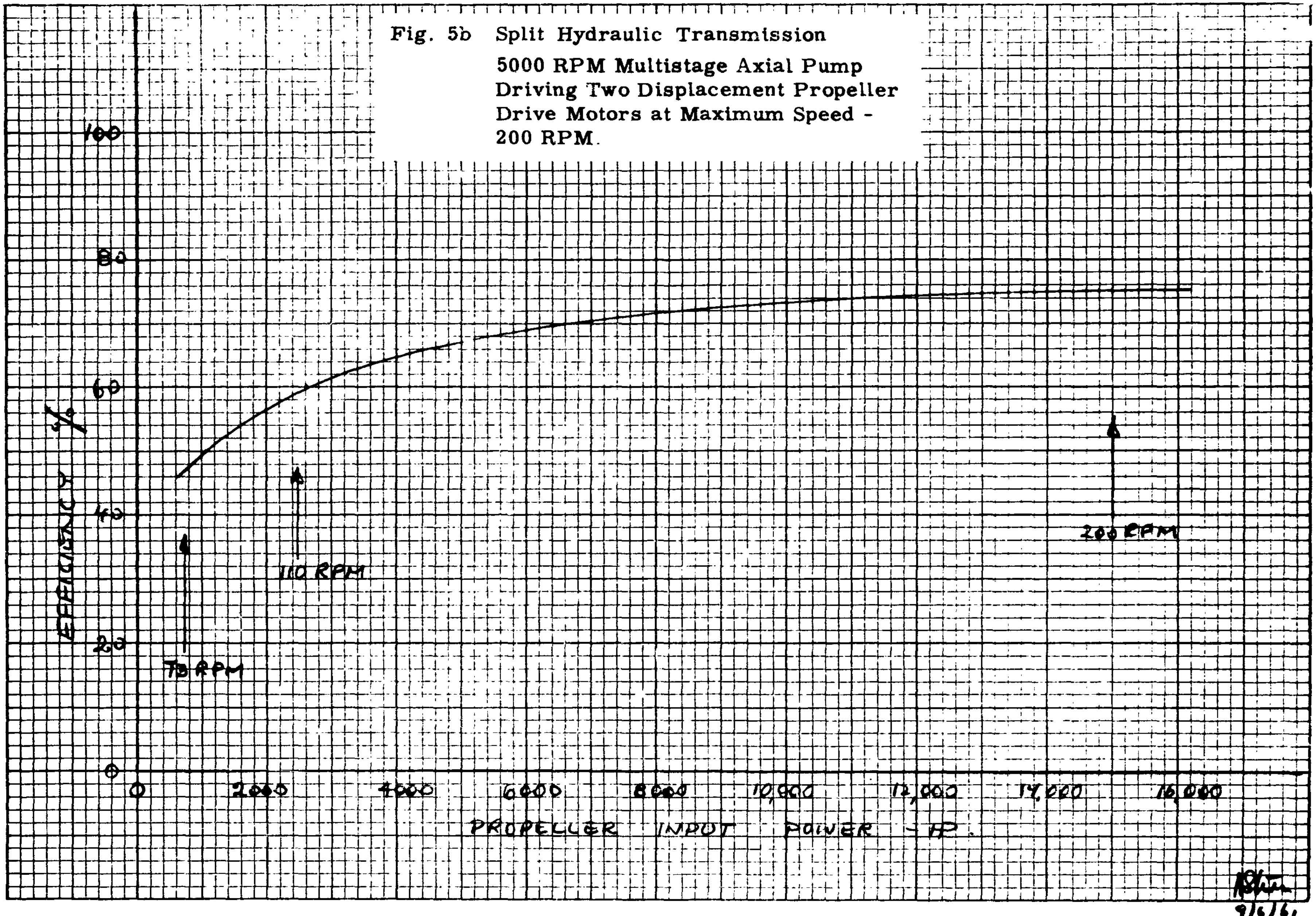
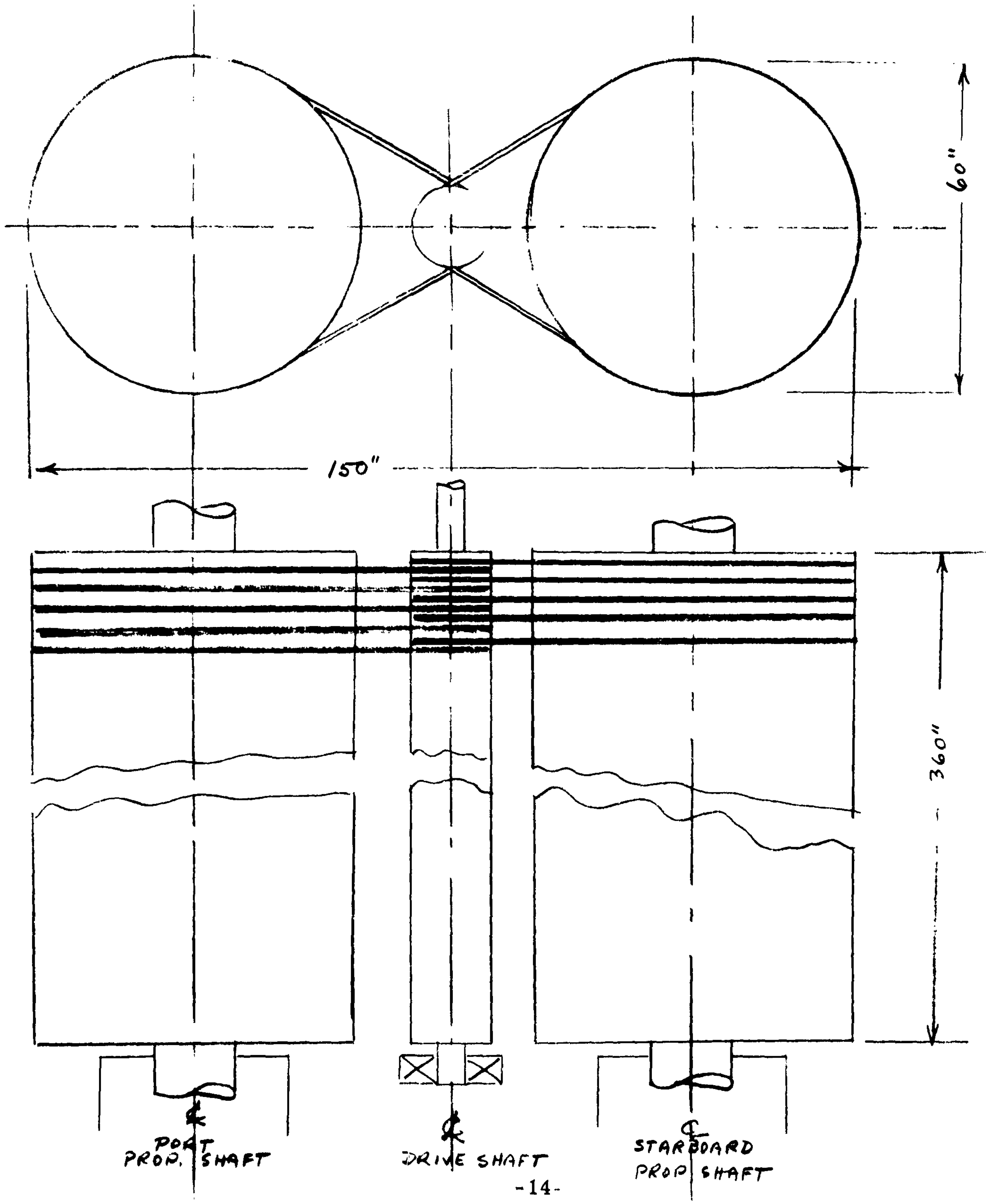


Fig. 5b Split Hydraulic Transmission
5000 RPM Multistage Axial Pump
Driving Two Displacement Propeller
Drive Motors at Maximum Speed -
200 RPM.

Fig. 6 Schematic of V-Belt or Rope Drive

Ratio - 4:1 or 1600 RPM - 400 RPM
Belts - 320 (total number)
Power - 15, 000 HP Max.



V belt bending causes the significant temperature rise. The chief design problem with a V belt drive is the fact that a 200 rpm propeller will not be capable of utilizing the most economic belt speeds. This puts the belt drive at an initial disadvantage for propeller drive applications. Normal design factors demand around 160 belts per 7500 hp propeller at 400 rpm propeller speed. With recent advances in belt design, this might be reduced to 100 belts. The geometry of a V belt transmission strongly favors a single turbine dual propeller drive, see Figure 6. The length of such a transmission in a axial direction can be expected to be far in excess of the equivalent gear transmission and other drives so far investigated. An approximate figure for transmission length along the turbine drive shaft is 360 inches. The transverse dimension of the transmission will be 162 inches and the height approximately 72 inches. The noise level of properly dry lubricated V belt drives is low if the belt speed is held to reasonable values such as 3 - 5,000 ft/min. In belt drives which are open to atmospheric air, wind noise generated at the sheaves may become significant particularly if air is used to cool the high speed sheave. In the case of the submarine transmission, it may be desirable to operate the V belt drive in the low pressure steam space of the condenser for both noise and cooling reasons. While reversing and variable pitch sheaves could be designed for such an application the mechanism appears to be too complex and space consuming to achieve these features with reliability. The most we, therefore, can expect from a V belt or rope drive is the equivalent in performance of a gear transmission at reduced noise level at the expense of size. Belt life will not be a serious problem at the power ratings demanded here since the transmission sketched in Figure 6 is sized for peak power. A severe limitation of this drive is its inability to bridge the large speed gap from high speed turbine to 200 rpm propeller in a single step. It is unthinkable to propose a double reduction sheave system when even the single step takes on 30 ft. in axial length of sheave alone.

3.4 Variable Pitch Propellers must be considered as transmissions since they perform one of the essential functions of a transmission, namely vary thrust power at constant rpm. In the frame of this study only a small amount of time was devoted to variable pitch propellers. A variable pitch propeller must be coupled to a speed reducer, like any other propeller, if a high speed turbine drive is used. Thus the complexity of the pitch varying mechanism is added to the 30:1 transmission, whatever that be. This step is not in keeping with the intent of this study program i.e., improving reliability of power transmission equipment. In a direct drive from a steam motor, the variable pitch propeller may become desirable to improve maneuvering characteristics of the ship. The same reasoning is used today in the application of variable pitch propellers on diesel powered tugs. Both diesel engines and steam motors exhibit similar speed-torque characteristics and would benefit from a variable ratio transmission such as the variable pitch propeller.

A version of the variable pitch propeller which may offer greater reliability for mechanical design reasons, is the ducted propeller with variable stator vanes. Such a propeller can, by varying the tangential component of the inlet and discharge flows approach the performance of a variable pitch runner. Investigations are in progress now exploring the potential of such a thrust generator. No information was made available to evaluate this device and little is found in published literature to permit such an evaluation. If a steam motor drive is used, the investigation of some variable geometry thrust generator will become necessary.

2.5 With a steam motor drive it may appear desirable to incorporate a 4:1 torque converter to permit higher motor operating speeds. This step should be taken with caution since a fluid torque converter can be expected to add considerably to the complexity of the system, and lower its efficiency. One of the desirable features of the steam motor, its reversibility will be lost since torque converters are unidirectional. Reversal of rotation will make a two element torque converter necessary, just as if the prime mover was unidirectional. Some improvement in speed/thrust characteristics may be attained by the use of a variable pitch propeller or a ducted propeller with variable stator geometry

3. Transmission Fluids

An important consideration in the selection of a transmission is the fluid used to cool, lubricate or transmit power.

In a steam turbine drive it would clearly be of advantage to be able to use condensate in the transmission bearings and as a power transfer fluid. Fottinger torque converters have used water as a transmission fluid in the past. Current automotive and marine practice is to use an oil. If water can be used in the bearings of the turbine it also can be used in the torque converter. Thus it is possible to eliminate a considerable amount of sealing, and supplant certain feedwater heaters by the torque converter which according to Reference 1 can give a $50^{\circ}F \Delta t$ to the boiler feed water. In this respect the development potential for the torque converters is better than that of any split hydraulic transmission using a positive displacement element. The necessarily high loading on the pistons and journals of such a machine eliminates water as a transmission fluid. Figure 5a shows among the components the oil/water heat exchanger which will reject the heat generated in the split hydraulic transmission to the feed water of the boiler. The sizing of such an exchanger is simple for hydrostatic transmission case since the high pressure-low flow characteristic does not require that special attention be paid to heat exchanger pressure drop. For torque converters or hydrodynamic transmissions the problem of heat exchanger size and losses becomes a major one.

Seawater was considered for a special low pressure hydraulic transmission whose drive motors are in a flooded compartment located in an annular space around the central torpedo tube (Ref. 8). This transmission eliminates all shaft penetrations of the pressure hull, excepting the pump shaft seal. Seawater is pumped by a high-speed turbine-coupled pump to two radial piston propeller drive motors in a closed circuit. Cooling water and make-up is taken on from the sea and rejected via the piston motors as leakage to the sea. Development of a seawater tolerant piston motor is a major obstacle in the exploitation of this design. Brief reference is made to this design in Section 2 above.

Another area where fluid is needed is in the control circuit of the power transmission. If hydraulic and lubricating oils are eliminated from the power train the use of water in the control circuit should also be considered. It is therefore desirable to pump all accessory fluids, such as transmission fluid, bearing lube fluid, and control fluid from a single pump. If this function can be combined with the steam generating circuit, a greatly simplified seal and piping arrangement can be obtained. The only mechanical transmission which holds out any hope for such a simplification is the Föttinger torque converter driven by a steam turbine or steam motor.

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Study of Hydraulic Transmission for Marine Propulsion

6/22/61 F. M. MacDonald
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G. E. Co., Lynn, Mass.

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Dec. 10, 1953 The National Supply Co.
Pittsburgh, Pa.

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Evaluation Test of 60" Marine Torque Converter

11 July 1958 U.S.N. Eng'g. Exp. Sta.
Annapolis, Md.

R 4 Same Part C
2 June 1959

R 5 Same Part D
14 March 1960

R 6 Full Circuit Calculated Performance of 597 Reversible Marine Converter

May 23, 1955 The Elliott Co.
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R 7 Sketch of Steam Driven Paddle Mechanism

June 16, 1961 R. J. Hooker
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R 8 Memorandum by H. Stern to Dr. B. Sternlicht, G. E. Co., Schenectady, N. Y.

7/19/61

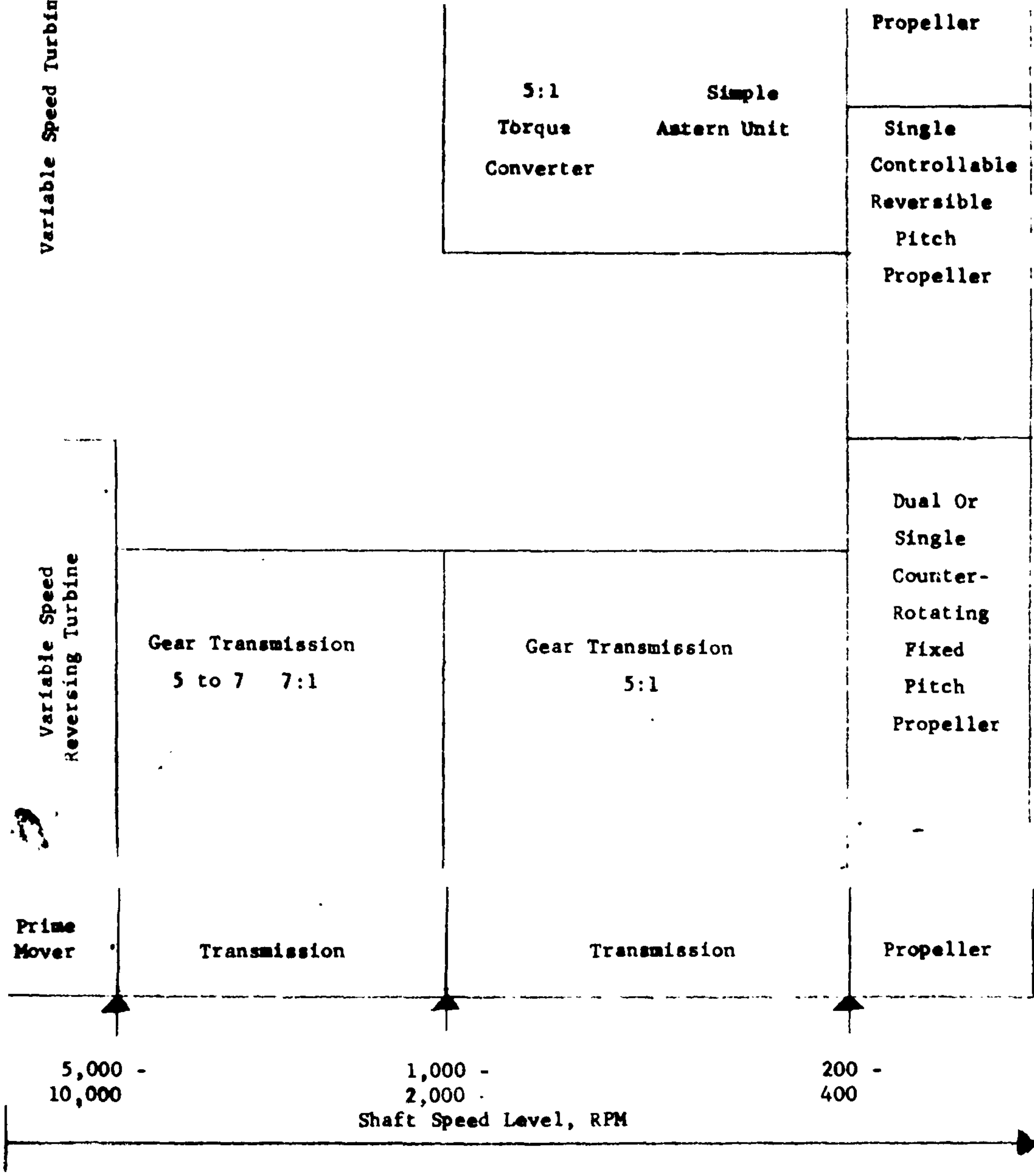
Multi-Cylinder Reciprocating Steam Motor (Reversible)		Single Fixed Pitch Propeller	28. 143,000 lb., 12' dia., 27' long - For efficiency, maneuverability, and quiet operation. This has greatest potential of all purely mechanical systems. If engine reliability can be made equal to turbine this becomes the optim. system.
	5:1 V - Belt Transmission (Fixed Geom.)		27. Only to be considered if low speed steam motor of (28) is <u>not</u> feasible. V-Belts are too heavy and space consuming.
Multi-Cylinder Reciprocating Steam Motor Non-Reversing	5:1 Simple Torque Converter	Single Controllable and Reversible Pitch Propeller	26. Adds unnecessary components to (28) just to get engine size down. Total space and weight will be greater and efficiency lower than (28). Ease of reversing steam motor must be traded for better maneuverability.
			25. Only advantage is greater maneuverability and slightly higher efficiency at part load. Dual controls are considerably more complex than (28).
	30:1 Hydrodynamic Shear Drive		24. Available information currently insufficient to evaluate qualitatively or quantitatively.
	30:1 D.C. Drive		23. Available information currently insufficient to evaluate qualitatively or quantitatively.

Constant Speed Steam Turbine (Non Reversing)		30:1 D.C. Drive		23. Available information currently insufficient to evaluate qualitatively or quantitatively.
Constant Speed Steam Turbine (Non Reversing)		30:1 A.C. Drive		22. Available information currently insufficient to evaluate qualitatively or quantitatively.
		30:1 Fluid Shear Transmission Variable Pitch And Reversing	Dual or Counter-Rotating Fixed Pitch Propeller	21. New development - high accuracy disc drive requires untried manufacturing technique and thermally well balanced operating conditions - not likely to be applicable to this program. Either tangential contact disks or periphera contact rollers are in this category

2

Constant Speed Steam Turbine (Non Reversing)	5:1 Torque Converter	5:1 Gear Transmission	Single Controllable and Reversible Pitch Propeller	20b. Same as 19
Constant Speed Steam Turbine (Non Reversing)	Gear Trans-Mission 5 to 7:1	5:1 Gear Transmission	Single Controllable and Reversible Pitch Propeller	20a. Same as 19
Constant Speed Steam Turbine (Non Reversing)	Gear Trans-Mission 5 to 7:1	5:1 Torque Converter (Fixed Geom.)	Single Controllable and Reversible Pitch Propeller	19. Offers no substantial advantage over 1 and 2.
Constant Speed Steam Turbine (Non Reversing)			Single Controllable and Reversible Pitch Propeller	18. 445,000 lbs. wet weight @ 15,000 HP requires hydraulic propeller pitch control. Low turbine efficiency, but no transmission loss quiet, except for propeller control.
Constant Speed Steam Turbine (Non Reversing)	5:1 Torque Converter	Fixed Geom. Simple	Single Controllable and Reversible Pitch Propeller	17. 320,000 lbs. Simplest T-C application, but requires controllable propeller - better maneuvering capability than 16.
Constant Speed Steam Turbine (Non Reversing)	5:1 Torque Converter	Var. Geom. and Astern Unit		16. 330,000 lbs. estimate safest application of T-C at Present. Efficiency will equal 18; quiet operation, best mechanical transmission.

Constant		Torque Converter	Var. Geom. and Astern Unit	Single Fixed Pitch Propeller	16. 330,000 lbs. estimate safest application of T-C at Present. Efficiency will equal 18; quiet operation, best mechanical transmission.
	Single Stage Axial Flow Pump (Var. Inlet Geom.)	7 Stage Axial Turbine (and Reversing Turbine			15. Unproven transmission design, low efficiency likely - uses condenser water except in pump controls - quiet operation see 7 and 8.
Constant Speed Steam Turbine (Non Reversing)	5 Stage Axial Flow Pump (Var. Inlet and Exit Geom.	2 Hydrostatic Propeller Drive Motors Reversible	Sea Water 1000 PSI Max.	Dual C/R Concentric Propellers	14. 215,000 lbs. Unlikely design to succeed. Reliability poor efficiency low noise level questionable.
			Hydraulic Fluid 2000 PSI Max.	Dual Fixed Pitch Blades	13. 175,000 lbs. uses separate hydraulic fluid for transmission and controls - noise questionable efficiency better than 15, equals 10.
	5:1 Torque Converter Var. Geom.	5:1Torque Converter Low Speed Var. Geom.	Astern Torque Converter	Single Fixed Pitch Propeller	12. 200,000 lbs. Could use water in T-C's at additional design risk. Lower efficiency than 18 due to tandem converters. Needs oil hydraulic controls complexity speaks for low reliability. High speed T-C is unexplored territory.
Variable Speed Turbine (Non Reversing)	5:1 Torque Converter Fixed Geom.	5:1 Torque Converter (Fixed Geom.)	Astern Torque Converter		11. 200,000 lbs. Has same properties as 12. Controls will be simpler since only turbine i controlled. Greater reliability than 12 - 15. Transmission efficiency cannot exceed 75%.
	5 Stage Axial Flow Pump (Fixed Geom.)	2 Hydrostatic Propeller Drive Motors Reversible	Hydraulic Fluid 2000 PSI Max.	Dual Fixed Pitch Propellers	10. 175,000 lbs. Best choice for light weig - reliability high, simple controls on turbin - efficiency of transmission alone. 75% at Max. power. Noise level of hydraulic motor in question.
			Sea Water 1000 PSI Max.	Dual C/R Concentric Propeller	9. 215,000 lbs. Same as 14 except simpler controls- sea water in hydrostatic Transmission.
	3	Separate 7 Stage			8. Same as 7, still lower efficiency, improved spacial arrangements possible. Trans



part load efficiency due to fixed prop. expect very low noise level except from turbine mounts.

5. 320,000 lbs. Reduces weight compared with 1 - 4 <u>but</u> add still more components and drops transmission efficiency to 80-85% with 90% hoped for. May be able to use feed water in T-C Quiet

4. 445,000 lbs. Adds Greater maneuvering ability but at expense of lower reliability due to propeller pitch controls. Poorer efficiency than 1-2 due to low speed turbine.

3. 445,000 lbs. Only current approach not using gears quiet only if prop. speed is held below 300 RPM which sacrifices turbine efficiency and weight.

2.

1. 350,000 lbs. Reliable, efficient, proven transmission system noise is undesirable - it comes from gears and variable speed turbine. Needs oil as gear lubricant.

<u>COMBINATIONS OF PRIME</u>
<u>MOVERS AND TRANSMISSIONS</u>
CONVERTING STEAM ENERGY INTO
PROPELLER SHAFT TORQUE

4

Table 1